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(54) **Variable-discharge high pressure pump**

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Description

BACKGROUND OF THE INVENTION

This invention relates to a variable-discharge high pressure pump (hereinafter sometimes referred to as "high pressure pump") according to the preamble of claim 1.

The preamble of the main claim proceeds from a variable-discharge high pressure pump of the kind described in the document EP-A-0 244 340.

According to this document a variable-discharge high pressure pump for use in a diesel engine comprises:

a plunger ;
a plunger chamber accommodating the plunger ;
a cam for reciprocatively moving the plunger wherein the cam includes a non-constant speed cam having a cam profile shaped such that the cam velocity of the cam is maximized in an initial stage of the forward stroking movement of the plunger ;
an electromagnetic valve capable of opening out to the interior of the plunger chamber ;
a fuel reservoir communicating with the plunger chamber through the electromagnetic valve ;
an inlet pipe for supplying a low pressure fuel to the fuel reservoir ; wherein
the introduction of the low pressure fuel from the inlet pipe into the plunger chamber and the return of the low pressure fuel from the plunger chambers to the inlet pipe are both effected through the electromagnetic valve ; and
a check valve communicating with the plunger chamber and capable of opening when the fuel pressure in the plunger chamber is raised beyond a predetermined pressure level.

One structural feature of this type of conventional high pressure pump resides in that a part of a low pressure fuel supplied through the inlet pipe is supplied to the reservoir while another part of the low pressure fuel is supplied to the plunger chamber. That is, a fuel inlet which opens into the plunger chamber and an outlet of the plunger chamber through which a part of the fuel is returned to the fuel reservoir are formed separately from each other. If in this high pressure pump the electromagnetic valve malfunctions by being fixed in a closed state, the flow of the fuel ejected through the check valve cannot be controlled. In such an event, there is a risk of the pressure in the common rail abruptly increasing and exceeding a limit pressure determined according to the strengths of the engine and the fuel injector and to the conditions for safety, resulting in damage to the members of the fuel injector.

SUMMARY OF THE INVENTION

It is therefore an object of the invention to further develop a high pressure pump of this genre such that, even if an electromagnetic valve malfunctions, the influx of fuel into the common rail at an excessively high pressure is preventable while the operability of the pump can be maintained with the least possible amount of energy being consumed.

According to the invention, this object is achieved by a high pressure pump having the features of claim.

The point of the invention to be stressed is the electromagnetic valve which can be operated by the pressure in the plunger chamber to a closed position. By this feature of the invention, when the pressure in the plunger chamber is sufficiently raised, the electromagnetic valve can be kept in the closed position after or even when the supply of the electric power to the electromagnetic valve is interrupted. Thus, the invention greatly reduces the consumption of the electric power.

When a reduced amount of fuel is to be discharged from the pump, the electromagnetic valve can be closed in a final stage of the forward stroking movement of the plunger. Accordingly, the electric power may be supplied to the electromagnetic valve for a shortened period of time, so that there is no problem in respect of the consumption of the electric power. On the other hand, when an increased or large amount of fuel is to be discharged from the pump, the electromagnetic valve must be closed from the initial stage of the forward stroking movement of the plunger. Thus, the electric power must be supplied to the electromagnetic valve for an extended time period, which results in the problem that the consumption of the electric power is increased. This problem is solved by the present invention.

In the present invention, a non-constant speed cam is employed to drive the plunger such that a high cam velocity is obtained in the initial state of the forward stroking movement of the plunger. By this feature of the invention, the electromagnetic valve is closed in the initial stage of the forward stroking movement of the plunger to cause a sharp increase in the fuel pressure in the plunger chamber. Thus, the fuel pressure in the plunger chamber can be raised in quite a short period of time to a pressure level high enough to keep the electromagnetic valve closed. Compared with the use of the conventional cam, therefore, the present invention can shorten the time period while the electric power must be supplied to the electromagnetic valve. In accordance with the present invention, therefore, the electromagnetic valve can be closed and kept in the closed position with a reduced consumption of the electric power even in the case where the electromagnetic valve is required to be closed from the initial stage of the forward stroking movement of the plunger.

These and other objects, arrangements and effects of the present invention will become more apparent upon reading the following description.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a longitudinal sectional front view of an embodiment of the present invention;

Fig. 2 is a longitudinal sectional view of the electromagnetic magnetic valve shown in Fig. 1;

Fig. 3 is a diagram of essential portions of the arrangement shown in Fig. 1;

Fig. 4 is a diagram of the construction of an engine fuel controller including the embodiment shown in Fig. 1;

Fig. 5 is a diagram of the electromagnetic valve opening and closing times and the plunger lift during ordinary control using reference pulses;

Fig. 6 is a flow chart of electromagnetic valve control relating a case where the return spring of the electromagnetic valve is broken;

Figs. 7 to 9 are diagrams showing a method of control for starting the engine;

Fig. 7 is a diagram showing a driving current supplied to the electromagnetic valve, the state of operation (opening/closing) of the electromagnetic valve corresponding to the driving current, the plunger displacement, and changes in the pressure in the plunger chamber;

Fig. 8 is a graph showing the relationship between the displacement of the plunger from the bottom dead point and the time required for the displacement;

Fig. 9 is a graph showing the relationship between the pump discharge Q and the difference T_T between the time at which the plunger lower dead point is reached and the time at which the electromagnetic valve is closed;

Fig. 10 is a longitudinal sectional front view of a part of variable-discharge high pressure pump which represents another embodiment of the present invention;

Fig. 11 is a graph of the cam velocity and the lift with respect to the cam angle;

Fig. 12 is a diagram of the operation of the pump shown in Fig. 11;

Fig. 13 is a front view of another example of the cam;

Fig. 14 is a graph of the cam velocity of the cam shown in Fig. 13 and the lift with respect to the cam angle;

Fig. 15 graphically illustrates the pressure characteristic of the common rail obtained when the fuel injection timing and the fuel pumping timing per unit of rotation are offset;

Fig. 16 graphically illustrates the pressure characteristic of the common rail obtained when the fuel injection timing and the fuel pumping timing per unit of rotation are registered; and

Fig. 17 is a longitudinal sectional front view of a conventional high pressure pump.

DESCRIPTION OF THE PREFERRED ENBODIMENTS

Referring to Fig. 1, a variable-discharge high pressure pump 10 which represents an embodiment of the present invention is illustrated. The high pressure pump 10 has a cam chamber 12 formed in a lower end portion of a pump housing 11, a cylinder 13 fitted in the pump housing 11, an inlet pipe 14 which is attached to the housing 11 and through which a low pressure fuel supplied from an unillustrated low pressure pump is introduced into the cylinder 13, and an electromagnetic valve 15 screwed into the cylinder 13.

A cam shaft 16 which rotates at a speed 1/2 of the rotational speed of the diesel engine extends through the cam chamber 12. A generally elliptical cam 17 is attached to the cam shaft 16. That is, while the diesel engine makes two revolutions to complete one cycle, the cam shaft 16 is driven to make one revolution.

The cylinder 13 has a slide hole 13a in which a plunger 18 is accommodated reciprocatively movably. The plunger 18 has a cylindrical shape and has no lead or the like. A plunger chamber 19 is defined by the plunger 18 and the slide hole 13a of the cylinder 13. A communication hole 21 is bored in the cylinder 13 so as to communicate with the plunger chamber 19. The inlet pipe 14 communicates with a fuel reservoir 22 formed between the cylinder 13 and the pump housing 11. The low-pressure fuel is supplied to the fuel reservoir 22 from the unillustrated low pressure pump through the inlet pipe 14.

A check valve 23 is attached to the cylinder 13 and communicates with the plunder chamber 19 through the communication hole 21. In the check valve 23, a valve plug 24 is forced to open the valve against a resultant force of the urging force of a return spring 25 and the fuel pressure in an unillustrated common rail by the fuel pressurized in the plunger chamber 19, thereby enabling the fuel to be ejected through an ejection hole 26 which communicates with the common rail via an unillustrated piping.

A spring seat 27 is connected to the plunger 18 at the lower end of the same. The spring seat 27 is pressed against a tappet 29 by a plunger spring 28. A cam roller 30 is rotatably attached to the tappet 29 and is brought into contact, under pressure, with the cam 17 disposed in the cam chamber 12 by the urging force of the plunger spring 28. The plunger 18 can therefore be moved reciprocatively by the cam roller 30 and the spring seat 27 which move in the longitudinal direction of the cylinder by following the contour 17a of the cam 17, as the cam shaft 16 rotates. The displacement and the speed of the reciprocative movement of the plunger 18 with respect to a certain rotational angle of the cam 17 are determined by the contour 17a of the cam 17.

The electromagnetic valve 15 is screwed into an lower end portion of the cylinder 13 so as to face the plunger 18. As shown in Fig. 2, the electromagnetic

valve 15 has: a body 32 in which low pressure passages 31 are formed so as to open at their inner ends into the plunger chamber 19; an armature 36 attracted in the direction of the arrow A of Fig. 2 against the urging force of a spring 35 (applied in the direction of the arrow B of Fig. 2) by the magnetic force of a solenoid 34 energized through a lead wire 33; and a mushroom valve plug 38 which is an opening-out valve capable of opening and closing the low pressure passages 31 by being moved integrally with the armature 36 to be fitted to or moved apart from a seat 37 formed at a plunger chamber 19 opening portion. The pressure of the fuel in the plunger chamber 19 is applied as a pressing force in the valve closing direction (in the direction of the arrow A of Fig. 2) to the valve plug 38. The electromagnetic valve 15 is a pre-stroke-control type of electromagnetic valve which serves to set the time at which pressurizing the plunger 18 is started by being energized at a predetermined time so as to fit the valve plug 38 to the seat 37. As shown in Fig. 1, the low pressure passages 31 communicate at their outer ends with the fuel reservoir 22 via a gallery 39 and a passage 40.

The embodiment is characterized in that the plunger chamber 19 and the inlet pipe 14 communicate with each other through the fuel reservoir 22 and the electromagnetic valve 15 alone, and both the introduction of the low pressure fuel into the plunger chamber 19 and the return of the low-pressure fuel to the fuel reservoir 22 are effected through the electromagnetic valve 15.

The difference between it and the conventional art will become more clear after examination of the construction of a conventional high pressure pump shown in Fig. 17. In Fig. 17, the same reference characters as those in Fig. 1 designate identical or equivalent portions or members, and the description for them will not be repeated.

As can be seen in Fig. 17, a conventional high pressure pump 10a is provided with feed holes 20 which communicate with the fuel reservoir 22, and the low pressure fuel is supplied to the fuel reservoir 22 through the inlet pipe 14 and the feed holes 20. Also, the low pressure fuel is supplied to the plunger chamber 19 through the inlet pipe 14 and the feed holes 20. That is, the feed holes 20 serving as a fuel inlet of the plunger chamber 19 and the low pressure passages 31 serving as an outlet for the return flow constitute different fuel passages. The feed holes 20 are opened or closed by the plunger 18, and the low pressure fuel is supplied to the plunger chamber 19 through the feed holes 20 when the feed holes 20 are not closed by the plunger 18. The high pressure pump thus constructed in accordance with the conventional art entails the problem of failure to control the pressure of the fuel if a valve accident takes place in which the valve plug 38 of the electromagnetic valve 15 is fixed in the valve closing state so that the pressure of the fuel ejected through the check valve 23 increases abruptly.

In accordance with the present embodiment, the feed holes 20 are eliminated and the low pressure passages 31 of the electromagnetic valve 15 also serve as a fuel supply passage, so that the fuel introduced into the fuel reservoir 22 is supplied to the plunger chamber 19 via the passage 40 formed in the cylinder 13, the gallery 39 and the low pressure passages 31 formed in the electromagnetic valve 15. Part of the fuel returns from the plunger chamber 19 to the fuel reservoir 22 by flowing in a direction opposite to the direction of the supply flow to the plunger chamber 19. In the thus-constructed pump, the supply of the fuel to the common rail is completely stopped if a valve accident takes place in which the valve plug 38 of the electromagnetic valve 15 is fixed in the valve closing state.

Fig. 3 schematically illustrates essential portions of the high pressure pump 10.

Referring to Fig. 4, the inlet pipe 14 of the high pressure pump 10 communicates with a fuel tank 4 through a low pressure passage 2 and a low pressure supply pump 3, and the ejection hole 26 of the check valve 23 communicates with a common rail 6 through a high pressure fuel passage 5. The common rail 6 is connected to injectors 7a to 7f corresponding to cylinders 8a to 8f of a diesel engine 1. A controller 9 is provided which has a CPU 9a, a ROM 9b, a RAM 9c and an input/output section 9d and which outputs valve opening/closing closing signals to the injectors 7a to 7f while being supplied with necessary data from the engine 1 and the common rail 6.

In this arrangement, during the downward movement of the plunger 18, the solenoid 34 of the electromagnetic valve 15 is not energized and the valve plug 38 is maintained in a valve opening state by the urging force of the return spring 35. The low pressure fuel supplied from the supply pump 3 therefore flows into the plunger chamber 19 via the inlet pipe 14, the fuel reservoir 22, the return outlet 31 of the electromagnetic valve 15 and the valve plug 38. At an initial stage of the upward movement of the plunger 18, the valve plug 38 is still in the opening state, and part of the fuel contained in the plunger chamber 19 is returned to the fuel reservoir 22 via the valve plug 38, the low pressure passages 31 and the gallery 39. If at this time the solenoid 34 is energized, the solenoid has an attraction force larger than the urging force of the return spring 35, thereby setting the valve plug 38 in a valve closing state. The fuel pressure in the plunger chamber 19 thereby increases. When this fuel pressure exceeds the sum of the urging force of the return spring 25 of the check valve 23 and the fuel pressure in the common rail 6, the check valve 23 opens to allow the fuel to be supplied under pressure to the common rail 6 through the high pressure passage 5. After this pressure feed has been completed, the energization of the solenoid 34 of the electromagnetic valve 15 is stopped, thereby setting the valve plug 38 in the valve opening state. The control of the high pressure pump 10 effected by energizing or de-energizing the solenoid 34 in synchronization with the rotation of the diesel en-

gine 1 on the basis of a signal from a sensor 100 for detecting the angular position of the cam 17 is hereinafter called as "ordinary control". During the ordinary control, the energization/non-energization times may be selected to change the pressure feed stroke of the plunger 18 and, hence, the fuel pressure in the common rail.

Fig. 5 shows an example of the lift H of the plunger 18 of the high pressure pump 10 with time during the ordinary control. An electromagnetic valve control signal represents a valve closing instruction a control time T_{F1} after the output of a reference pulse. At this time, the plunger 18 has already been lifted to a predetermined extent. When the electromagnetic valve 15 is closed, the pressure feed of the fuel from the high pressure pump is started, thereby supplying the mount of fuel corresponding to a stroke defined between this lift and the full lift H_{max} (H_1 shown in Fig. 5) to the common rail 6 under pressure.

If the signal for closing the electromagnetic valve 15 is issued a control time T_{F2} after the reference pulse, the lift of the plunger 18 determined at this time is large, and the pressure feed stroke is correspondingly small as defined by H2. Thus, the pressure feed amount is reduced if the control time is increased, or the pressure feed amount is increased if the control time is reduced. It is therefore possible to control the pressure feed amount by selecting the time at which the electromagnetic valve 15 closing signal is issued.

Even if during the operation of the high pressure pump 10 the electromagnetic valve 15 is fixed in the closed state, and if the plunger 18 is moved downward in this state, the fuel supplied to the electromagnetic valve 15 from the supply pump 3 does not flow into the plunger chamber 19. Accordingly, when the plunger 18 is moved upward, the fuel is not Supplied to the common rail under pressure, and there is no possibility of the injector 7 being damaged.

In a case where the return spring 35 loses the force of urging the valve plug 38 by, for example, being broken, the valve plug 38 is moved to open the valve by the effect of the difference between the pressures in the gallery 39 and the plunger chamber 19 as the plunger 18 is moved downward, thereby allowing the fuel supplied to the electromagnetic valve 15 from the supply pump 3 to flow into the plunger chamber 19. As the plunger is thereafter lifted, the pressure in the plunger chamber 19 becomes higher than the pressure in the gallery 39. At this time, the valve plug 38 is moved to close the valve since the return spring 35 has no urging force, and the fuel inside the plunger chamber 19 is pressurized and is supplied to the common rail 5 through the check valve 23 under pressure. That is, the fuel is supplied to the common rail 6 under pressure even if the solenoid 34 of the electromagnetic valve 15 is energized. The pressure in the common rail 6 is thereby abruptly increased, there is therefore a risk of damage to the members of the fuel injector.

Fig. 6 shows a flow chart of a method of preventing this risk. In the process of Fig. 6 involving the ordinary control, if the rate at which the pressure in the common rail changes becomes positive during the non-energized state of the solenoid 34, it is determined that an abnormality of the electromagnetic valve 15 takes place, and the solenoid 34 is continuously maintained in the energized state. The signal indicating that the pressure change rate is positive can be obtained by the calculation of a signal from a pressure sensor 6a provided in the common rail 6, which calculation is performed by the controller 9. The controller 9 outputs the valve closing signal to the electromagnetic valve 15. In this control process, the electromagnetic valve 15 is maintained in the closed state, thereby preventing the fuel from flowing into the plunger chamber 19 of the high-pressure pump 10 and, hence, from being supplied to the common rail under pressure.

Figs. 7 to 9 are diagrams of a method of abruptly increasing the pressure in the common rail 6 when the engine is started by using the high pressure pump in accordance with this embodiment.

At the time of starting, the engine rotates at a low speed, and, if the electromagnetic valve 15 is controlled in the ordinary control manner, it takes a long time to increase the pressure in the common rail 6 due to lack of voltage for the CPU 9a or lack of output from the cam 17 angle sensor 100. To avoid this problem, as shown in Fig. 7, pulse signals asynchronous with the revolutions of the high pressure pump 10 and having an energization time T_1 and a non-energization time T_2 are applied to the electromagnetic valve 15. The valve plug 38 is moved to close the valve a valve closing delay time T_c after the start of energization and is moved to open the valve a valve opening delay time T_0 after the start of non-energization. The plunger 18 is moved upward during the time when the valve plug 38 is in the valve closing state, thereby increasing the pressure in the plunger chamber 19.

The valve plug 38 is of the opening-out type, and is maintained in the valve closing state even when the solenoid 34 is not energized, once the pressure P_k in the plunger chamber 19 becomes higher than the valve closing maintenance pressure P_1 of the valve plug 38. The valve closing maintenance pressure P_1 is expressed by the following equation using the load F_s of the return spring 35, the diameter D_s of the seat of the valve plug 38, the supplied fuel pressure P_1, and π:

$$P_1 = \frac{F_s}{\pi \cdot D_s^2/4} + P_f$$

During the valve closing maintenance state of the valve plug 38, the pressure in the plunger chamber 19 is increased as the plunger 18 is moved upward, thereby supplying the fuel to the common rail 6 through the check valve 23 under pressure.

After plunger 18 has been moved downward so that the pressure in the plunger chamber 19 becomes lower than the valve closing maintenance pressure P_1 of the valve plug 38, the valve plug is moved so as to repeat the valve opening/closing operations by the pulse current flowing through the solenoid 34. Thus, during the valve opening state of the valve plug 38, the fuel flows into the plunger chamber 19 via the valve plug 38.

The setting of the energization time T_1 and the non-energization time T_2 in accordance with this pulse control will be explained below.

The energization time T_1 is obtained which is required to produce, during the minimum speed rotation for starting the engine, the pressure in the plunger chamber 19 to maintain the valve plug 38 in the valve closing state, after the plunger 18 of the high pressure pump 10 has started moving upward from the bottom dead point. The average lifting displacement AH of the plunger 18 for producing the valve closing maintenance pressure P_1 can be obtained by the following equation using the supplied fuel pressure P_f, the fuel capacity V, the bulk modulus E of the fuel, the diameter D_k of the plunger, and π:

$$\Delta H = \frac{(P_1 - P_f)\cdot V}{E\cdot\pi\cdot D_k^{\,2}/4}$$

As shown in Fig. 3, a limit of the fuel capacity V is defined at the seat of the check valve 23 provided that the check valve 23 opening pressure is larger than the valve closing maintenance pressure P_1 of the valve plug 38.

The time ΔT required to displace the plunger 18 by ΔH is maximized at the plunger bottom dead point, as shown in Fig. 8. Let the time ΔT required to displace the plunger 18 by ΔH from the bottom dead point during the minimum rotation for starting the engine be T_3, and the valve closing time delay for the operation of the valve plug 38 be T_c. Then, the energization time T_1 is expressed by the following equation:

$$T_1 = T_3 + T_c$$

In accordance with fuel drawing conditions, the non-energization time T_2 is set to enable the maximum fuel discharge Q_{max} to be drawn during one valve opening period, as expressed by the following equation:

$$T_2 = \frac{Q_{max}}{C\cdot S\cdot\sqrt{P_f - P_k}} + T_0 - T_c$$

where C represents a constant determined by physical properties including the viscosity of the fuel, and S represents the flow passage area.

In Fig. 9, the solid line indicates the pump discharge Q mm^3/st with respect to the difference T_T between the time at which the plunger 18 is positioned at the bottom dead point and the time at which the electromagnetic valve 15 is closed. If in this case the pulse control period $(T_1 + T_2)$ is doubled, the pump discharge changes as indicated by the broken line, that is, the change in the discharge Q becomes larger and the average discharge becomes reduced. Accordingly, it is possible to reduce the change in the discharge Q while increasing the average discharge by reducing the period $(T_1 + T_2)$, thereby enabling the pressure in the common rail 6 to be increased faster. The energization time T_1 and the non-energization time T_2 for pulse control are determined on the basis of this examination.

Referring then to Fig. 10, a high pressure pump 10c which represents an embodiment of the present invention is illustrated in section. In this embodiment, a cam 17b has a generally elliptical cam profile defined by concave circular-arc cam surfaces 17c and other curved cam surfaces 17d. Assuming that the point in the cam profile corresponding to the bottom dead point of the plunger 18 defines a cam angle of 0°, the curved surface 17c is formed between cam angles of 0° and about 30° with a curvature of R_1 the center of which is outside the cam 17b. The center of curvature of the surfaces 17d is inside the cam 17b. The plunger 18 reaches the to dead point at a cam angle of 90°. Because a portion of the cam profile corresponding to an initial stage of the up stroke is defined by the concave circular-arc surface 17c, the speed of upward movement of the plunger 18 is accelerated by the cam surface at this stage. Fig. 11 shows a graph of the cam velocity and the lift with respect to the angle of the cam 17b. As the cam angle is increased, a peak of the cam velocity is exhibited when the cam angle and the lift are small. As the cam angle is further increased until the dead point is reached, the cam velocity decreases. The rate at which the lift is increased is larger at a stage where the cam angle is small, i.e., during the period of time corresponding to the first half of the up stroke where the lift is small. The lift increasing rate is smaller during the period of time corresponding to the second half of the up stroke where the lift is large and the cam velocity is decreasing. The cam 17b effects up-down strokes two times during one revolution of the cam shaft 16 and exhibits a non-constant-velocity cam curve such that the lifting speed is gradually increased during the first half of lifting and is reduced during the second half of lifting.

Next, the operation of the variable-discharge high pressure pump in accordance with this embodiment will be explained below with respect to time with reference to Fig. 12. An electromagnetic valve control signal represents an instruction for valve closing for a time T_D a control time T_{L1} after the output of a reference pulse from the cam angle sensor 100. At this time point a, the plunger 18 has been moved upward to a lift P_1. The electromagnetic valve 15 is closed at the time point A to start supplying the fuel under pressure. The amount of fuel

corresponding to a part S_1 of the stroke defined between this time point A and a time point C at which the plunger 18 reaches the highest point P_3 is thereby discharged into the common rail. In a case where the electromagnetic valve control signal represents a valve closing instruction a control time T_{L2} after the reference pulse (as indicated by the broken line), i.e., at a time point B, the lift of the plunger 18 at this time point is P_2 and pressure feed of the fuel is only effected with a part S_2 of the stroke between a height P_2 and a height P_3. That is, the amount of fuel supplied to the common rail under pressure is reduced if the control time T_L after the reference pulse is increased, or is increased if the control time T_L is reduced. It is therefore possible to control the discharge by selecting the control time T_L.

Next, the relationship between the cam velocity, the control time and the plunger lift will be examined below.

Since in this embodiment the cam velocity is set to be higher for the first half of the up stroke of the plunger, the cam velocity changes with respect to time as indicated by the solid line in Fig. 12. That is, in a case where the control time T_{L1} is short and the discharge is large, the cam velocity at the time point A at which pressure feed is started (when the valve is closed) is V_1 and increases as the pressure feed proceeds. The cam velocity exhibits a peak during the period of time corresponding to the first half of the up stroke of the plunger, and thereafter decreases gradually.

Then, the pressure feed state in the case where the cam velocity is set so as to be higher during the period of time corresponding to the second half of the plunger up stroke will be examined below for comparison with the pressure feed in the case of the variable-discharge high pressure pump in accordance with this embodiment. If the peak of the cam velocity is set for the second half, the change in the cam velocity with time is as indicated by the double-dot-dash line in Fig. 12; the cam velocity at the control start time point A is Vx. As can be understood from the graph, the cam velocity Vx is lower than the cam velocity V_1 at the control start time point A in the case of this embodiment.

The control signal represents the electromagnetic valve closing instruction after the control time T_{L1} from the reference pulse, and allows valve Opening after a period of time T_D.

Even when valve opening is allowed by the signal and when the electromagnetic valve is in the non-energized state, the electromagnetic valve is maintained in the closed state by the pressure in the plunger chamber if this pressure is high, since the electromagnetic valve of the variable-discharge high pressure pump in accordance with the present invention is of the opening-out type. The pressure feed is therefore continued until the plunger to dead point is reached. However, during low-speed operation or, more specifically, during the operation in a super-low-speed range for starting the engine in which a large discharge is required to promptly produce and maintain the common rail pressure, the plunger lifting speed is, in fact, lower even if the same cam profile is used, resulting in a reduction in the pressure increase rate. On the other hand, the valve closing setting time T_D is minimized because it is desirable to reduce the valve closing time T_D, i.e., to establish the valve opening allowance state faster in order to enable the variable-discharge high pressure pump to be used for operation of a higher speed. In such a case where the cam velocity is low while the valve closing time T_D is short, the fuel pressure in the plunger chamber does not increases to a level sufficient for maintaining the closed state of the electromagnetic valve, and the valve is opened before the pressure feed to be continued until the dead point is reached is completed, thereby allowing the fuel to return to the fuel chamber. As a result, the discharge becomes naught although the signal designates the large discharge.

However, in the case of the variable-discharge high pressure pump in accordance with this embodiment, the cam velocity is peaked for the first half of the plunger up stroke and, specifically, a certain acceleration is reached immediately after the control start point. The upward movement of the plunger is thereby accelerated so that the plunger moves at a high speed. At the initial stage of plunger lifting, therefore, the pressure in the plunger chamber can be increased in a short time to a level high enough to maintain the opening-Out type electromagnetic valve in the closed state. Thus, even if the valve closing setting time T_D is set to be shorter in order to enable the variable-discharge high pressure pump to operate suitably even at a high speed, it is possible to set, in the short valve closing setting time T_D, the pressure in the plunger chamber to a level high enough to maintain the closed state of the valve. It is thereby possible to continue the pressure feed until the plunger to dead point is reached and, hence, to ensure a large discharge during super-low-speed operation even though the valve opening allowance state is established after a short time.

In a case where a large discharge is not required, that is, an instruction to close the electromagnetic valve is issued with a control time T_{L2} delay, the cam velocity exhibited at the time point B as indicated by the solid line in Fig. 12 in the case of the cam for setting the peak for the first half of the up stroke is lower than that exhibited as indicated by the double-dot-dash line in Fig. 12 in the case of the cam for setting the peak of the cam velocity for the second half. In the case of the former type of cam, however, the pressure in the plunger chamber can be boosted more easily by the effect of the approaching period (T_{L2}) for opening the electromagnetic valve as well as the effect of reduction in the dead volume, and the internal pressure for maintaining the electromagnetic valve in the closed state can be obtained, thereby preventing the valve from opening again.

Thus, the variable high pressure pump in accordance with this embodiment is capable of ensuring a large discharge required during the super-low-speed

operation for, for example, starting the engine while satisfying requirements for high speed operation, thereby enabling the optimum common rail pressure to be produced stably irrespective of the operating conditions.

In accordance with a still another embodiment, a non-constant-velocity cam for creating strokes during one revolution of the cam shaft is used in place of the non-constant-velocity cam for creating two strokes during one revolution of the cam shaft in the variable-discharge high pressure pump in accordance with the above-described embodiment.

A cam in accordance with this embodiment will be described below with reference to Figs. 13 and 14.

Fig. 13 is a front view of a cam 132 whose profile is as described below. It is assumed that the point in the cam profile corresponding to the bottom dead point of the plunger 18 defines a cam angle of 0°. The corresponding cam surface is formed as a concave surface 133, and a crest 134 in the cam profile corresponding to the top dead point of the plunger 18 is formed at a cam angle α of 60°. The concave cam surface 133 has a circular-arc contour having a curvature R_2 the center of which is outside the cam 132, and is defined between cam angles of 0 and 20°. Another concave surface 133 is formed through an angle β between cam angles of about 100 and 120°. The rest of the cam surface in the range of these angles is formed as a curved surface 135 having a curvature the center of which is inside the cam 132. That is, the concave circular-arc surfaces 133 correspond to the first half of the up stroke and the second half of the down stroke, and the cam velocity is increased during the periods corresponding to these halves of the strokes. The cam 132 has other cam surfaces formed in the same manner; the crests 134 and the concave surfaces 133 are formed in three places so that the cam 132 exhibits three identical profile portions during one revolution of the cam shaft 16.

Fig. 14 is a graph showing the cam velocity of the cam 132 and changes in the lift with respect to the cam angle.

The cam velocity is peaked at about a cam angle of 20° for the first half of the up stroke. During the period of time corresponding to the first half of the up stroke, the lift is small but the lift increasing rate is large. During the period of time corresponding to the second half of the up stroke where the cam velocity decreases under the peak, the lift is large but the lift increasing rate is small.

That is, the cam 132 ensures that the fuel pressure can be increased to a high pressure by the first half of the up stroke. A variable-discharge high pressure pump in which the cam 132 is used has the same performance and effects as the above-described embodiments while the rotational speed of the cam shaft 16 is lower.

When an 8-cylinder Diesel engine is equipped with three high pressure pumps each operative to discharge fuel three times per rotation of a cam shaft, as shown in Fig. 13, i.e., per unit of rotation according to a cycle of the engine, the injector associated with each of the engine cylinders performs one injection, i.e., a total of eight injections by eight injectors, per unit of the engine rotation while the fuel is discharged and pumped into the common rail three times by each pump, i.e., a total of nine times by the three pumps, as will be seen from the curves named "Pumping Pressure" in Fig. 15.

Accordingly, because the cycle of the fuel injecting operations of the injectors is not registered with the cycle of the fuel discharges by the high pressure pumps, the pressure in the common rail is varied in the manner shown by the waves named "Imaginary Common Rail Pressure" in Fig. 15. Hummerings take place when the fuel injectors are closed, as shown by the waves named "Hummering Components" in Fig. 15. The hummerings are combined with the variation in the common rail pressure caused due to the fuel injections by the injectors and the fuel discharges and pumpings by the pump, so that the actual common rail pressure is varied in the matter shown by the waves named "Actual Common Rail Pressure" in Fig. 15. The variation of the actual common rail pressure shown in Fig. 15 is greatly smaller than the common rail pressure variation obtained when the timings of fuel injections by injectors are in registry with the timings of fuel discharges by the high pressure pumps, as shown in Fig. 16.

In the example discussed above, the fuel is injected through injectors into the engine eight times per unit of rotation while the fuel is discharged and fed into the common rail nine times per unit of rotation. In general, however, the variable-discharge high pressure pump may discharge the fuel into the common rail $\underline{n}$ times per unit of rerotation, the number $\underline{n}$ being equal to the number of injections by the injectors multiplied or divided by a non-integral number.

Claims

1. A variable-discharge high pressure pump (10) for use in a diesel engine (1), comprising:

    (a) a plunger (18);

    (b) a plunger chamber (19) accommodating said plunger (18);

    (c) a cam (17) for reciprocatively moving said plunger (18), wherein said cam (17) includes an non-constant speed cam having a cam profile shaped such that the cam velocity of said cam (17) is maximized in an initial stage of the forward stroking movement of said plunger (18);

    (d) an electromagnetic valve (15) capable of opening out to the interior of said plunger chamber (19);

(e) a fuel reservoir (22) communicating with said plunger chamber (19) through said electromagnetic valve (15);

(f) an inlet pipe (14) for supplying low pressure fuel to said fuel reservoir (22); wherein the introduction of the low pressure fuel from said inlet pipe (14) into said plunger chamber (19) and the return of said low pressure fuel from said plunger chamber (19) to said inlet pipe (14) are both exclusively effected only through said electromagnetic valve (15);

(g) a check valve (23) communicating with said plunger chamber (19) and capable of opening when the fuel pressure in said plunger chamber (19) is raised beyond a predetermined pressure level;

characterized in that
said electromagnetic valve (15) is operable by the pressure in said plunger chamber (19) towards a closed position, wherein fuel is fed under high pressure through said check valve (23) to a common rail (6) for an accumulation of the high pressure fuel to be injected through injectors (7a-7f); and that

(h) a control means is provided which is operative such that the supply of the electric power to said electromagnetic valve (15) is commenced at a predetermined time point during the forward stroking movement of said plunger (18) caused by said cam (17) to close said electromagnetic valve (15) so that the supply of the fuel from said plunger chamber (19) to said common rail (6) is commenced by the closing of said electromagnetic valve (15) at said predetermined time point and is finished by the completion of the forward stroking movement of said plunger (18); and
said control means is operative to supply the electric power to said electromagnetic valve (15) to close the same during only the time period until said electromagnetic valve (15) is closed and kept in the closed position by the pressure in said plunger chamber (19), whereby, after the pressure in said plunger chamber (19) is sharply increased by the operation of said non-constant speed cam, said electromagnetic valve (15) is kept closed by the pressure in said plunger chamber (19).

Patentansprüche

1. Hochdruckpumpe mit veränderlichem Abfluß (10) für die Verwendung in einem Dieselmotor (1), welche die folgenden Bauteile hat:

    a) einen Kolben (18);
    b) eine Kolbenkammer (19), die den Kolben (18) aufnimmt;
    c) eine Nocke (17) für die Hin- und Herbewegung des Kolbens (18), wobei die Nocke (17) eine nicht konstante Geschwindigkeits-Nocke mit einem Nockenprofil umfaßt, welches derart geformt ist, daß die Nockengeschwindigkeit der Nocke (17) in einem anfänglichen Stadium der Vorwärtshubbewegung des Kolbens (18) maximal ist,
    d) ein elektromagnetisches Ventil (15), welches für ein sich Öffnen hin zum Innenraum der Kolbenkammer (19) vorgesehen ist,
    e) ein Kraftstoffreservoir (22), das mit der Kolbenkammer (19) über das elektromagnetische Ventil (15) verbunden ist,
    f) eine Einlaßleitung (14) für das Zuführen von Niederdruckkraftstoff in das Kraftstoffreservoir (22), wobei das Einströmen von Niederdruckkraftstoff aus der Einlaßleitung (14) in die Kolbenkammer (19) und der Rückstrom von dem Niederdruckkraftstoff aus der Kolbenkammer (19) in die Einlaßleitung (14) jeweils ausschließlich durch das elektromagnetische Ventil (15) bewirkt wird,
    g) ein Rückschlagventil (23), das mit der Kolbenkammer (19) verbunden ist und dazu in der Lage ist, sich zu öffnen, wenn der Kraftstoffdruck in der Kolbenkammer (19) über ein vorbestimmtes Druckniveau ansteigt,

dadurch gekennzeichnet, daß
das elektromagnetische Ventil (15) durch den Druck in der Kolbenkammer (19) in Richtung einer Schließstellung betätigbar ist, wobei Kraftstoff unter hohem Druck durch das Rückschlagventil (23) zu einer gemeinsamen Leitung (6) für ein Ansammeln an Hochdruckkraftstoff gefördert wird, welches durch Einspritzeinrichtungen (7a bis 7f) eingespritzt werden soll und das

h) ein Regelmittel vorgesehen ist, welches derart betätigbar ist, daß mit der Zufuhr von elektrischer Energie zum elektromagnetischen Ventil (15) begonnen wird in einem vorbestimmten Zeitpunkt während der Vorwärtshubbewegung des Kolbens (18) verursacht durch die Nocke (17), um das elektromagnetische Ventil (15) zu schließen, so daß mit der Zufuhr an Kraftstoff aus der Kolbenkammer (19) zu der gemeinsamen Leitung (6) durch das Schließen des elektromagnetischen Ventils (15) in dem vorbestimmten Zeitpunkt begonnen wird und durch das Vollenden der Vorwärtshubbewegung des Kolbens (18) beendet wird, wobei das Regelmittel betätigbar ist, um die elektrische Energie an das elektromagnetische Ventil (15) anzulegen, um dieses während lediglich der Zeitperiode zu schließen, bis das elektromagnetische Ventil (15) geschlossen ist und in der geschlossenen Stellung durch den Druck in der Kolbenkammer (19) gehalten wird, wodurch

das elektromagnetische Ventil (15) durch den Druck in der Kolbenkammer (19) in geschlossenem Zustand gehalten wird, nachdem sich der Druck in der Kolbenkammer (19) durch den Betrieb der nicht konstanten Geschwindigkeits-Nocke erheblich erhöht hat.

Revendications

1. Pompe haute pression (10) à débit variable destinée à l'utilisation dans un moteur diesel (1), comprenant :

   (a) un piston (18) ;
   (b) une chambre de piston (19) logeant le piston (18) ;
   (c) une came (17) destinée au mouvement de va et vient du piston (18), la came (17) ayant une vitesse non constante et un profil de came tel que la vitesse de la came (17) est maximale à l'étape initiale du mouvement de course avant du piston (18) ;
   (d) une électrovanne (15) pouvant déboucher à l'intérieur de la chambre de piston (19) ;
   (e) un réservoir de carburant (22) communiquant avec la chambre de piston (19) par l'électrovanne (15) ;
   (f) un tuyau d'arrivée (14) pour amener le carburant basse pression au réservoir de carburant (22) ; l'introduction du carburant basse pression en provenance du tuyau d'arrivée (14) dans la chambre de piston (19) et le retour du carburant basse pression en provenance de la chambre de piston (19) vers le tuyau d'arrivée (14) s'effectuent tous deux exclusivement par l'intermédiaire de l'électrovanne (15) ;
   (g) un clapet anti-retour (23) communiquant avec la chambre de piston (19) et pouvant s'ouvrir lorsque la pression du carburant dans la chambre de piston (19) est portée au-delà d'un niveau de pression prédéterminé ;

   caractérisée en ce que

   l'électrovanne (15) est actionnable par la pression dans la chambre de piston (19) vers une position fermée, le carburant étant amené sous haute pression par le clapet anti-retour (23) à un rail commun (6) destiné à l'accumulation du carburant haute pression à injecter par les injecteurs (7a-7f) ; et en ce que
   (h) il est prévu des moyens de commande fonctionnant de façon que l'alimentation de la puissance électrique à destination de l'électrovanne (15) commence à un instant prédéterminé pendant la course avant du piston (18) provoquée par la came (17) pour fermer le clapet anti-retour (15) de sorte que l'alimentation de carburant en provenance de la chambre de piston (19) vers le rail commun (6) est mise en route par la fermeture de l'électrovanne (15) à un instant prédéterminé et se termine à l'achèvement du mouvement de course avant du piston (18) ; et
   le moyen de commande est opérant pour fournir la puissance électrique à l'électrovanne (15) pour fermer celle-ci uniquement pendant le laps de temps précédant la fermeture de l'électrovanne (15) et le maintien de cette dernière en position fermée par la pression dans la chambre de piston (19), moyennant quoi, après une brusque augmentation de la pression dans la chambre de piston (19) sous l'action de la came à vitesse non constante, l'électrovanne (15) est maintenue fermée par la pression dans la chambre de piston (19).

FIG. 1

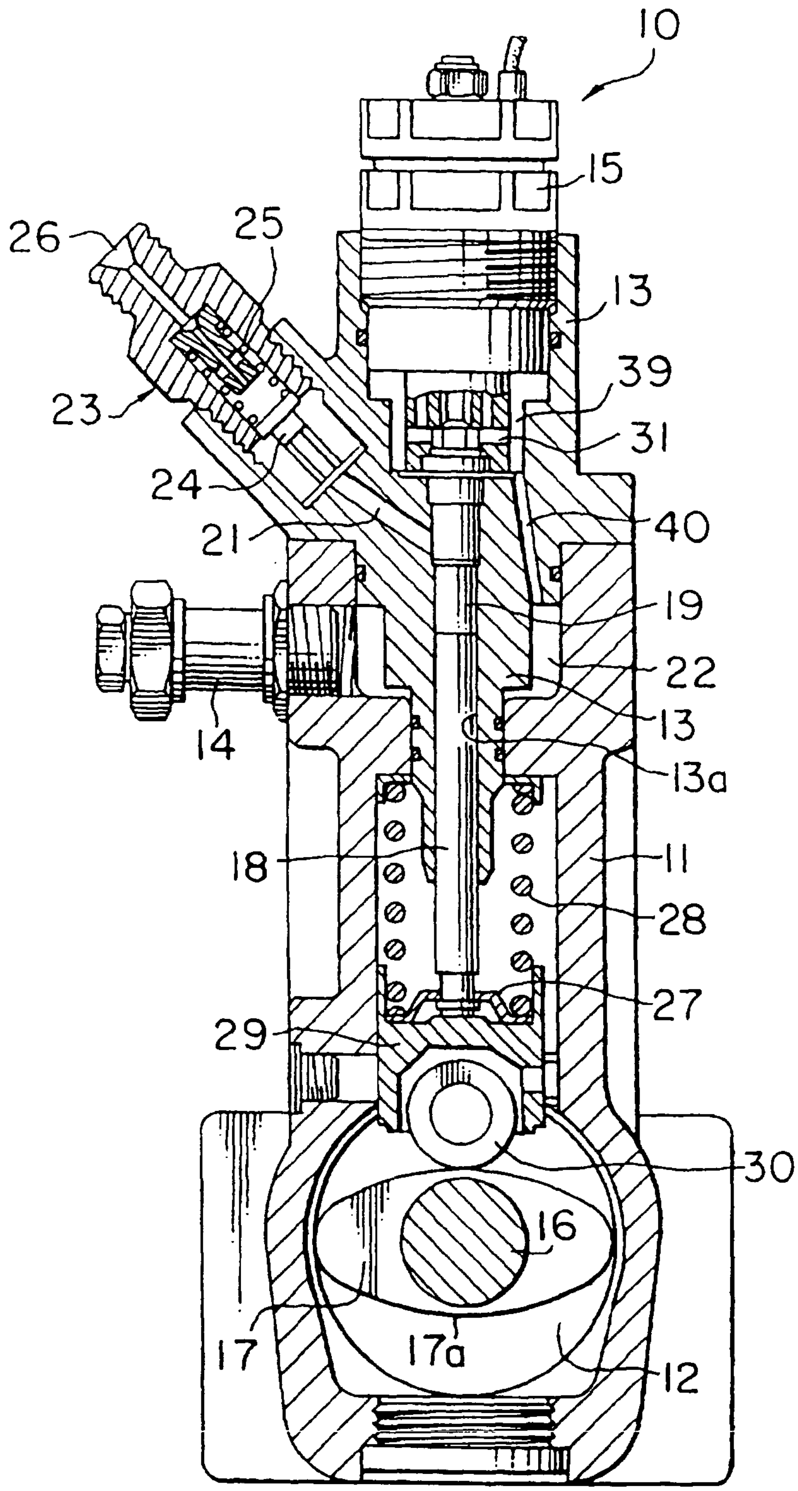

10
15
26
25
13
39
23
31
24
40
21
19
22
14
13
13a
18
11
28
27
29
30
16
17
17a
12

FIG. 2

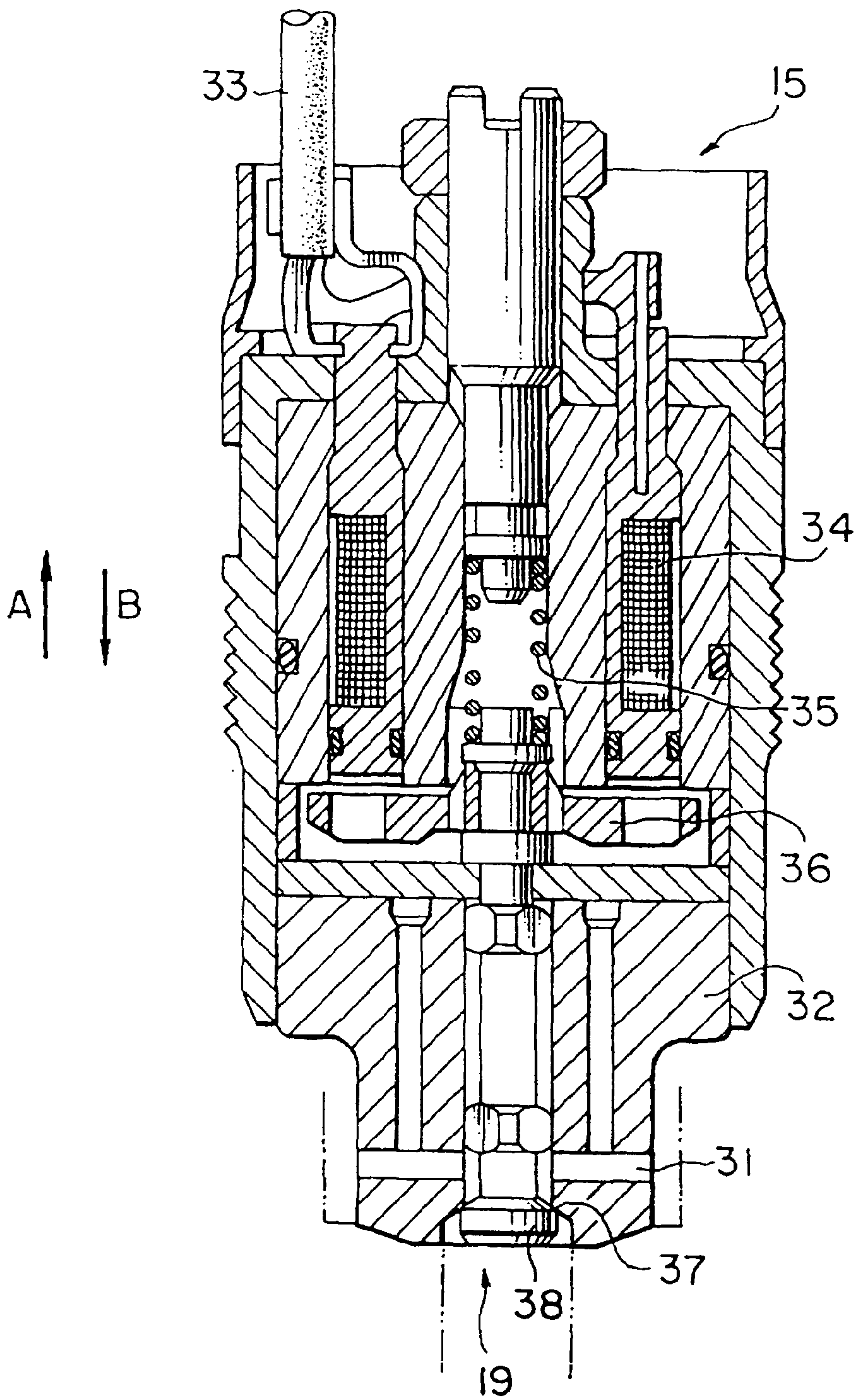


FIG. 3

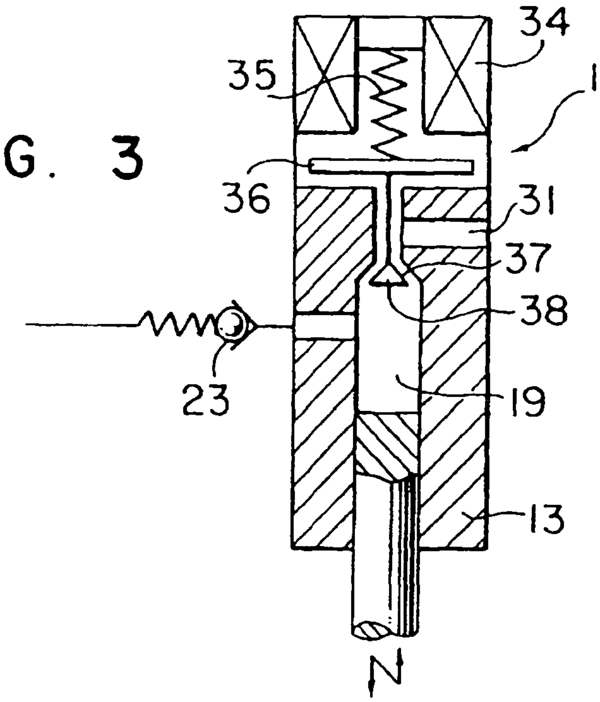

34
35
10
36
31
37
38
23
19
13
N

FIG. 4

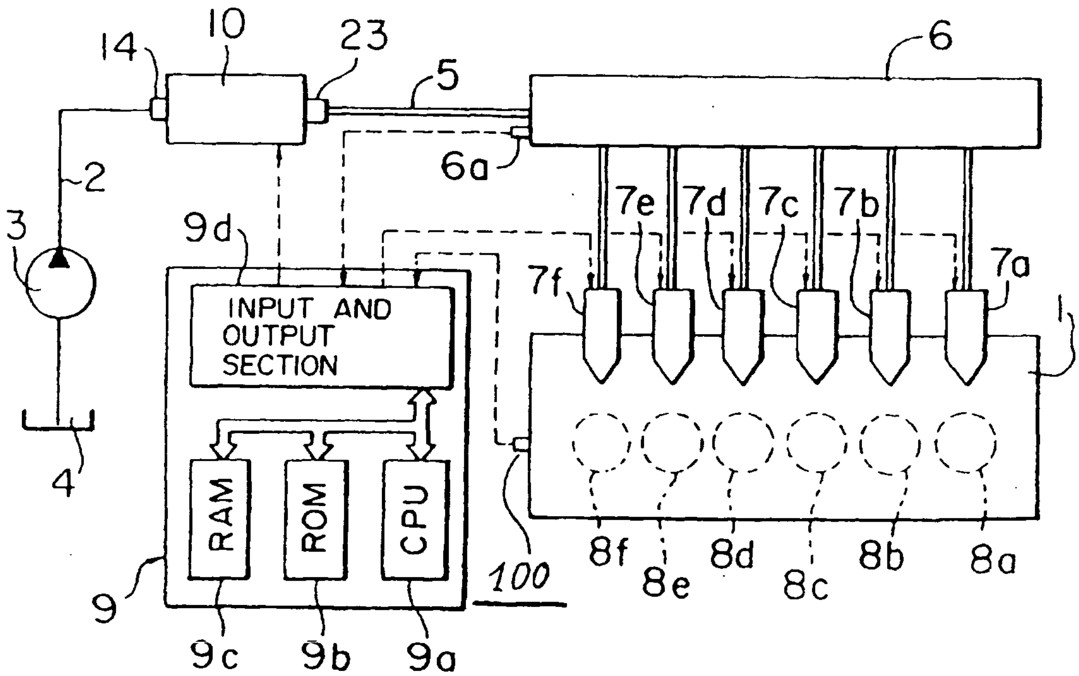

14
10
23
5
6
6a
2
3
9d
7e
7d
7c
7b
7a
7f
INPUT AND
OUTPUT
SECTION
1
4
RAM
ROM
CPU
100
9
8f
8e
8d
8c
8b
8a
9c
9b
9a

FIG. 5

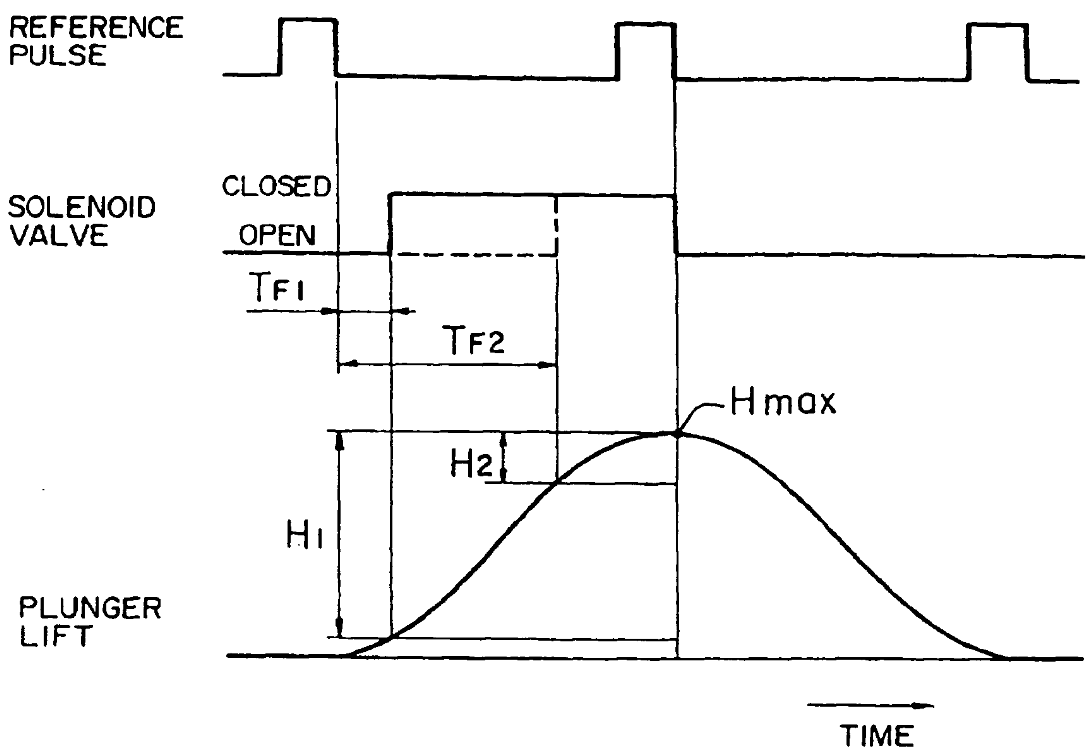

REFERENCE PULSE
SOLENOID VALVE
CLOSED
OPEN
TF1
TF2
Hmax
H2
H1
PLUNGER LIFT
TIME

FIG. 6

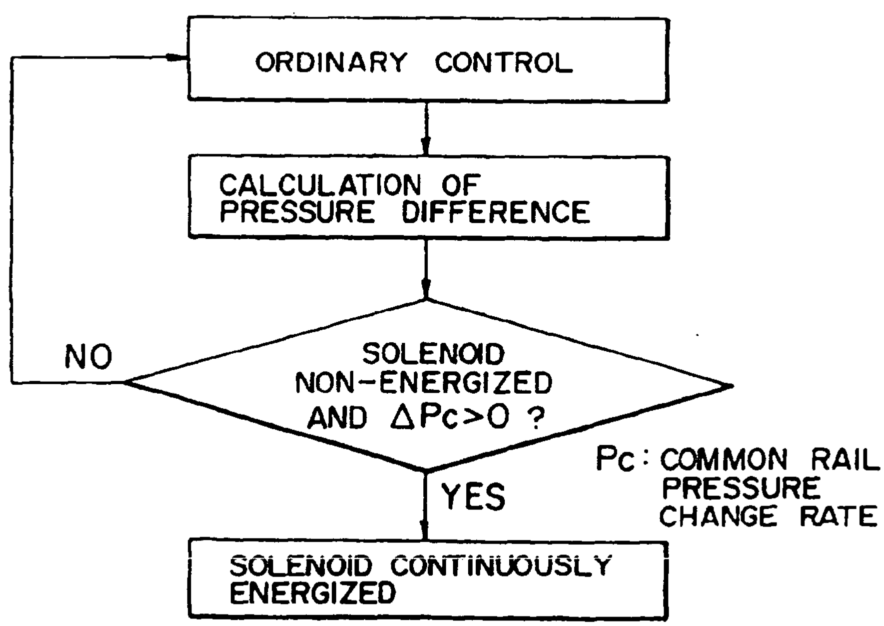

ORDINARY CONTROL
CALCULATION OF PRESSURE DIFFERENCE
SOLENOID NON-ENERGIZED AND ΔPc>0 ?
NO
YES
Pc: COMMON RAIL PRESSURE CHANGE RATE
SOLENOID CONTINUOUSLY ENERGIZED

FIG. 7

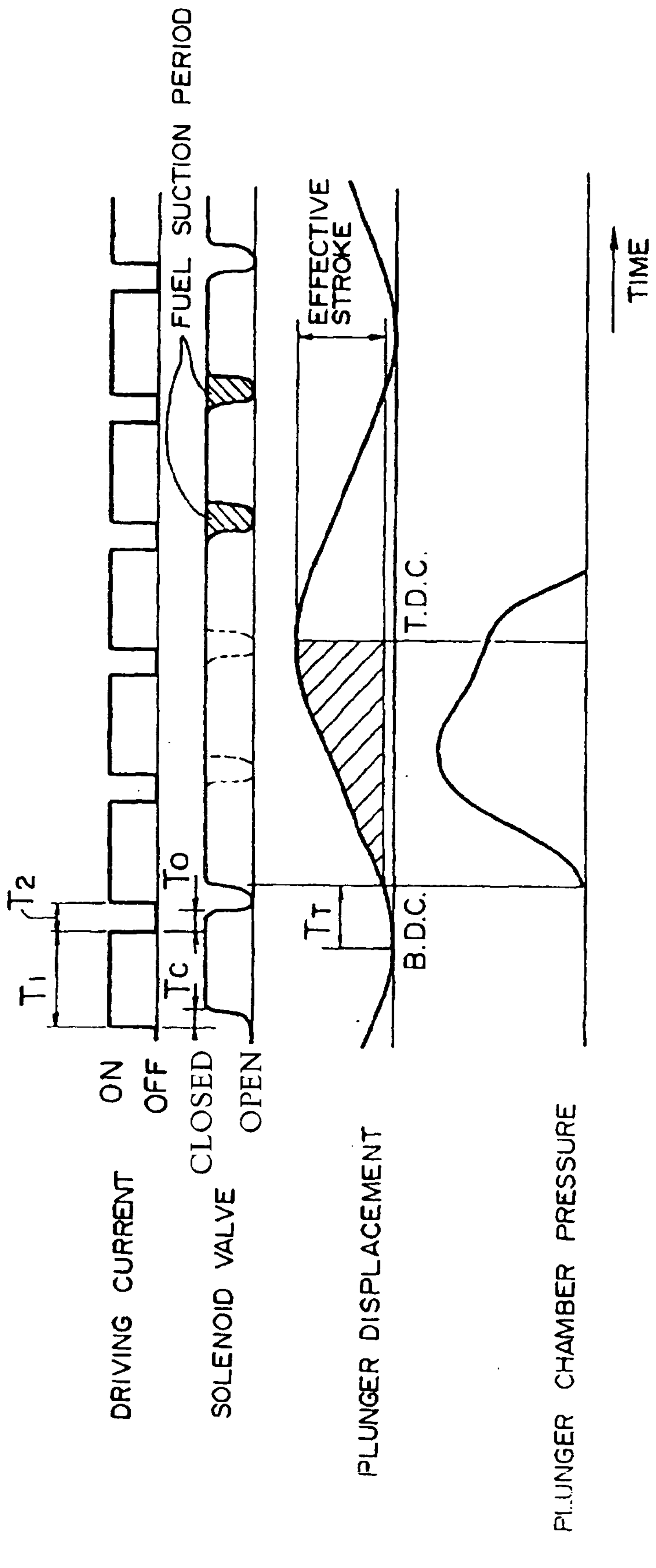

DRIVING CURRENT
ON
OFF
SOLENOID VALVE
CLOSED
OPEN
PLUNGER DISPLACEMENT
PLUNGER CHAMBER PRESSURE
T1
T2
Tc
To
TT
B.D.C.
T.D.C.
FUEL SUCTION PERIOD
EFFECTIVE STROKE
TIME

FIG. 8

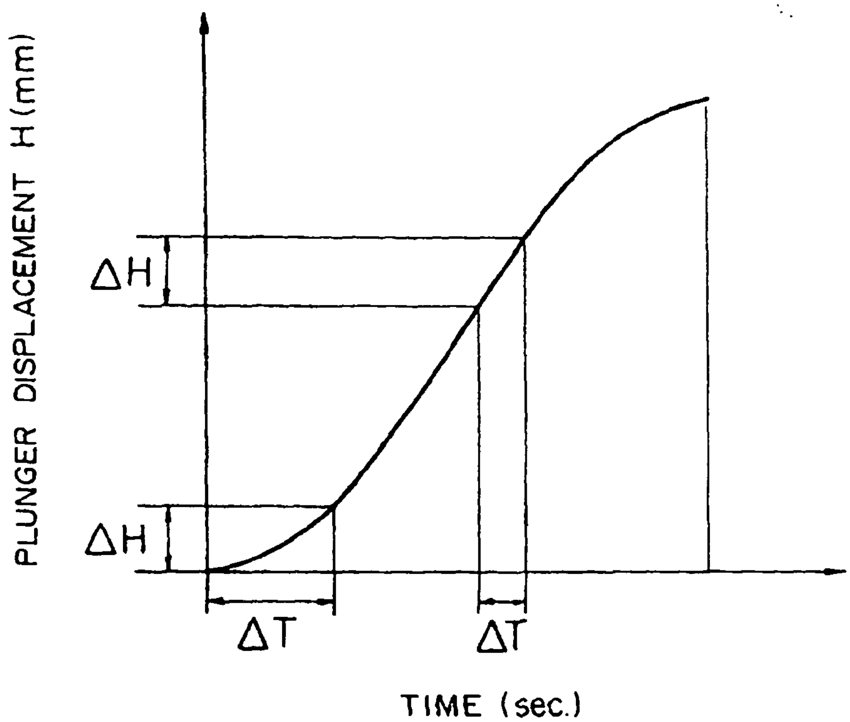

PLUNGER DISPLACEMENT H (mm)
ΔH
ΔH
ΔT
ΔT
TIME (sec.)

FIG. 9

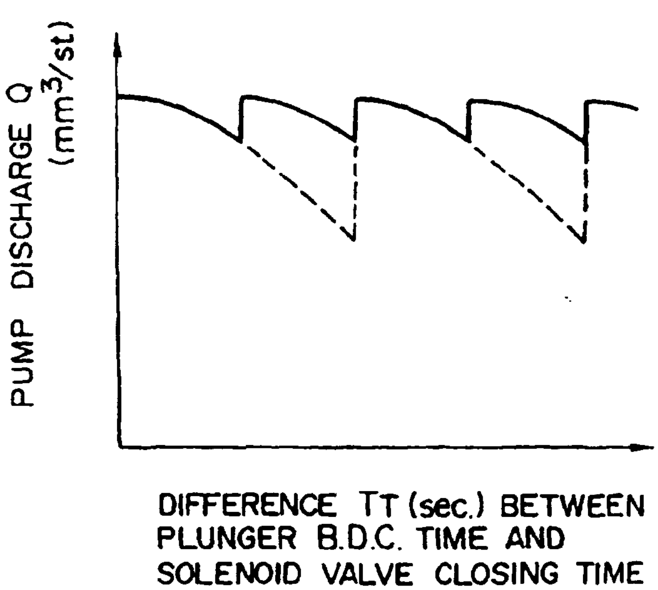

PUMP DISCHARGE Q (mm3/st)
DIFFERENCE TT (sec.) BETWEEN PLUNGER B.D.C. TIME AND SOLENOID VALVE CLOSING TIME

FIG. 10

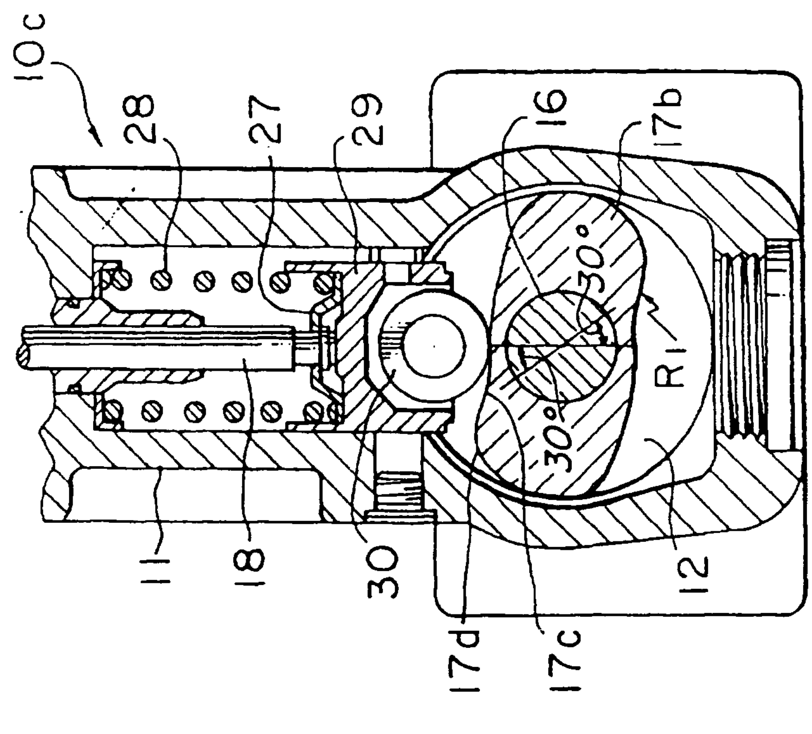

10c
28
27
29
16
17b
30°
30°
30°
R1
11
18
30
12
17d
17c

FIG. 11

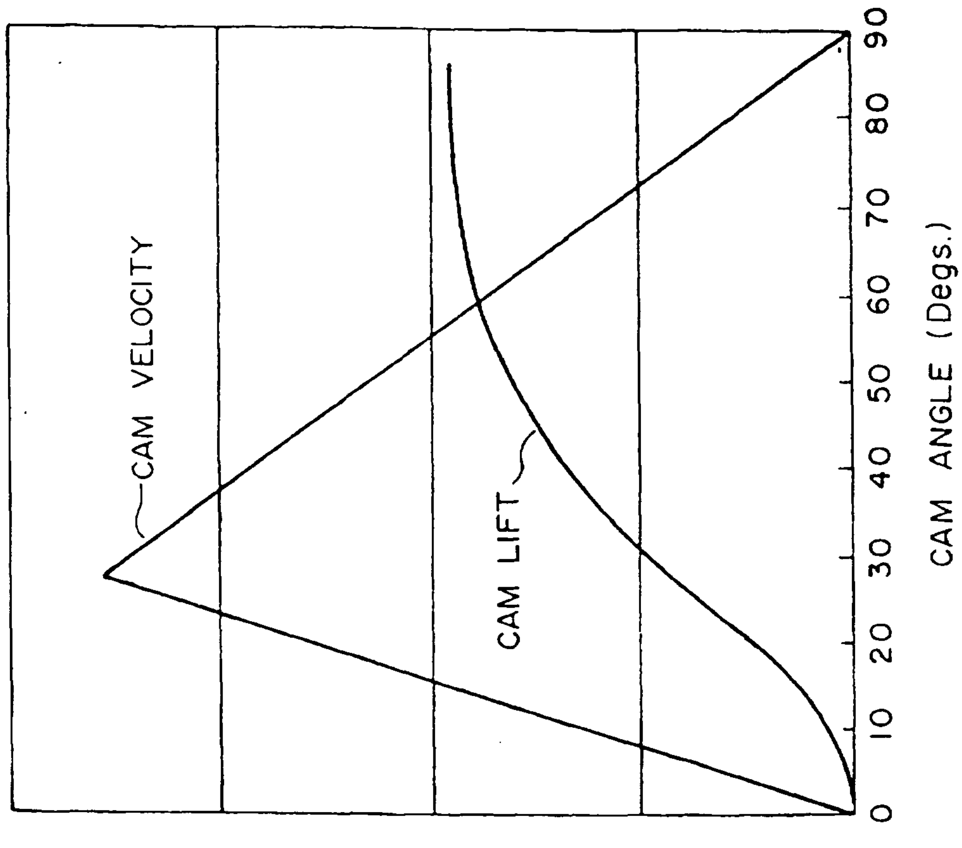

CAM VELOCITY
CAM LIFT
0
10
20
30
40
50
60
70
80
90
CAM ANGLE (Degs.)

FIG. 12

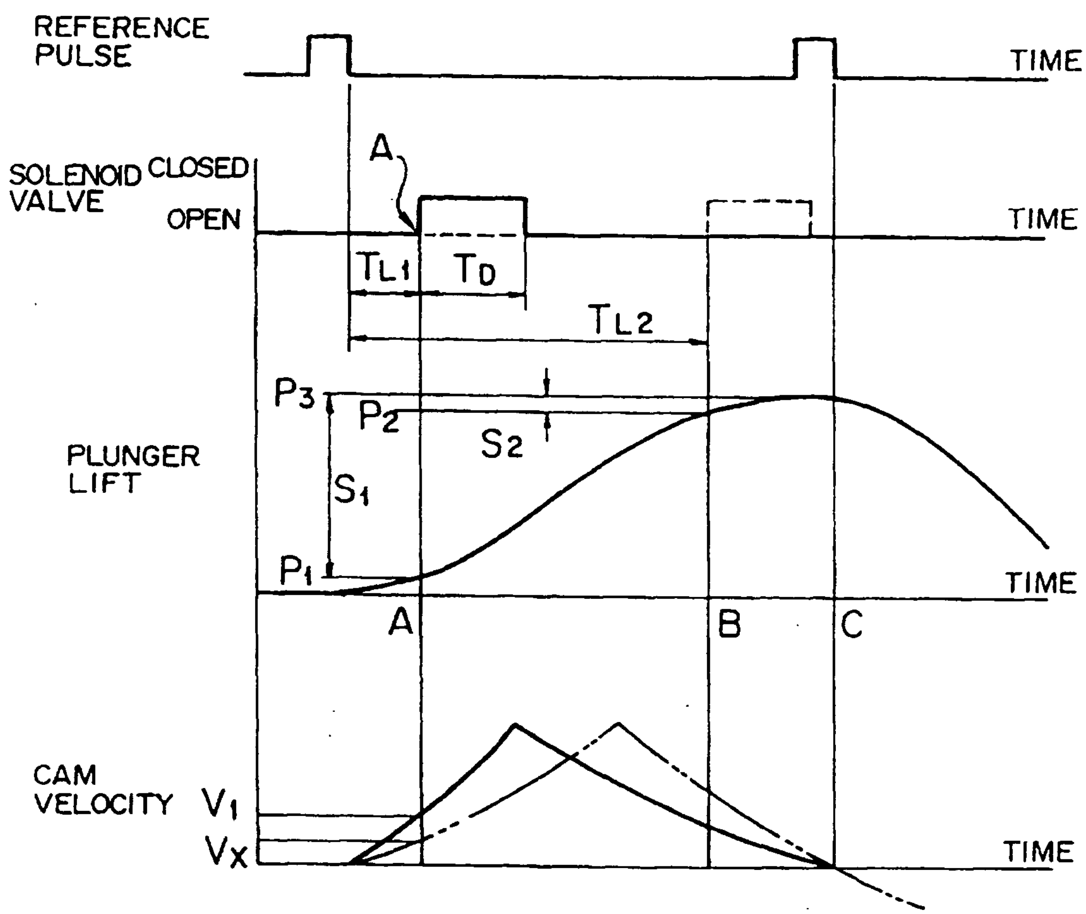

REFERENCE PULSE
TIME
SOLENOID VALVE
CLOSED
OPEN
A
TIME
TL1
TD
TL2
P3
P2
S2
PLUNGER LIFT
S1
P1
TIME
A
B
C
CAM VELOCITY
V1
Vx
TIME

FIG. 13

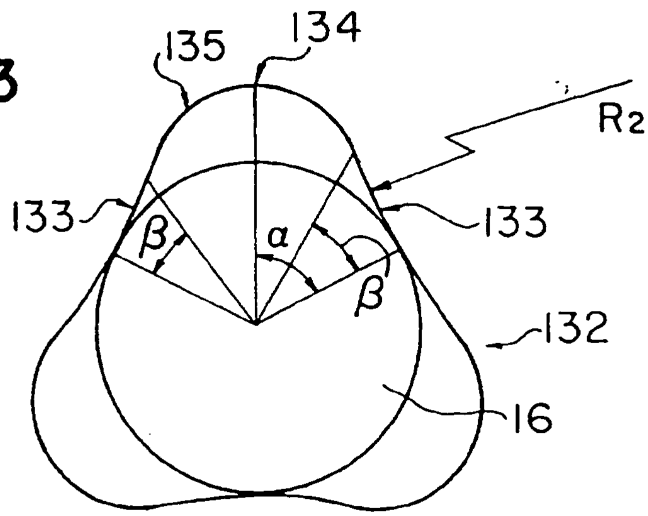

135
134
R2
133
133
β
α
β
132
16

FIG. 14

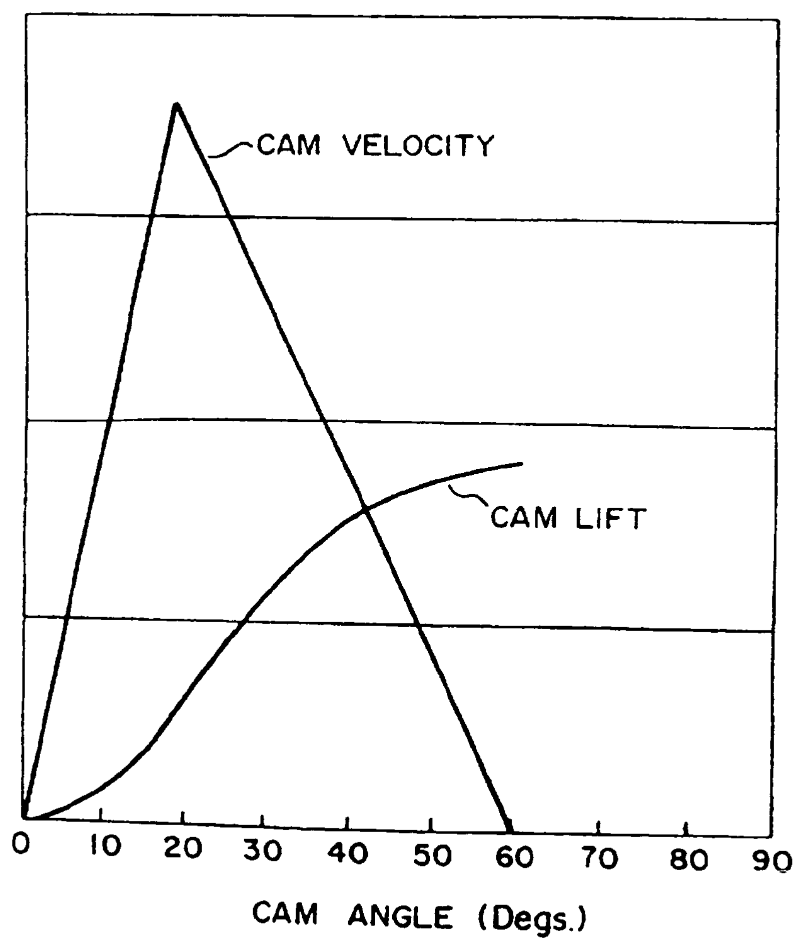

CAM VELOCITY
CAM LIFT
0
10
20
30
40
50
60
70
80
90
CAM ANGLE (Degs.)

FIG. 15

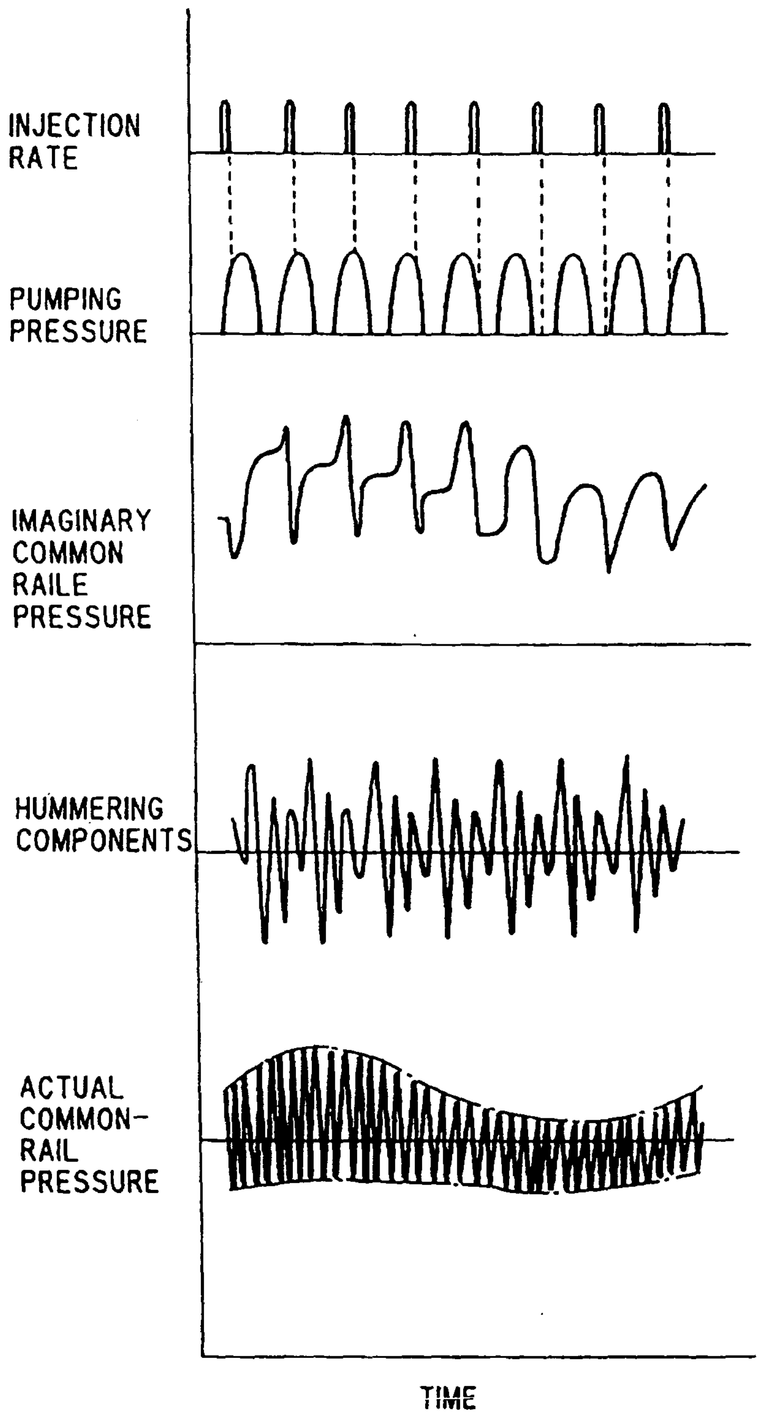

INJECTION RATE
PUMPING PRESSURE
IMAGINARY COMMON RAILE PRESSURE
HUMMERING COMPONENTS
ACTUAL COMMON-RAIL PRESSURE
TIME

FIG. 16

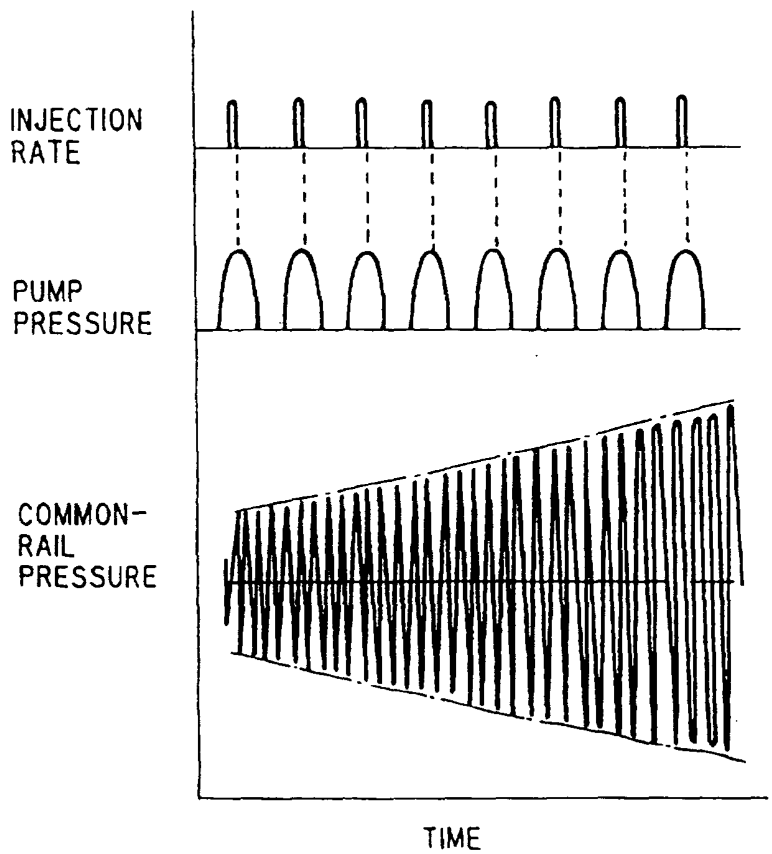
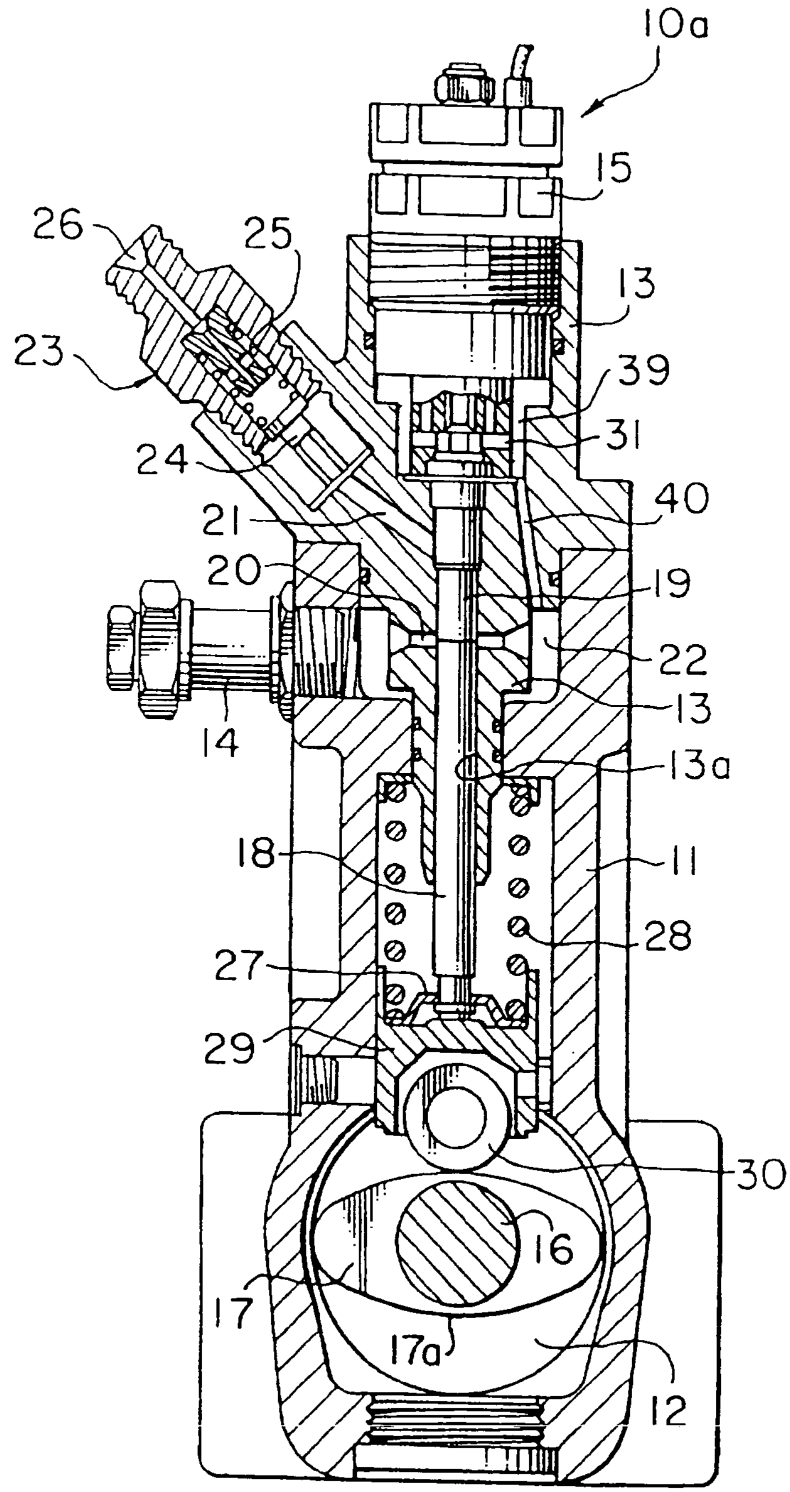


FIG. 17


10a
15
26
25
13
23
39
31
24
40
21
20
19
22
13
14
13a
18
11
28
27
29
30
16
17
17a
12